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European Domestic Heat Pumps with Capacity Control

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EUROPEAN DOMESTIC HEAT PUMPS WITH CAPACITY CONTROL

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ABSTRACT

The use of heat pump systems for domestic heating in Northern Europe is characterized by long function periods with moderate ambient temperatures during which the system will run on partial load. Under these conditions, the system can be effectively improved by adapting capacity to actual demand. By computer simulation, different grades of capacity modulation are described and compared. As an example of such a compressor, the newly-introduced twin compressor for domestic heat pump systems is described.

INTRODUCTION

In Northern Europe a significant interest in heat pumps for domestic heating has arisen in recent years and there are now large numbers of models on the market. There were initial difficulties with heat pumps, but an acceptable level of reliability and performance has been reached. Looking ahead however, if the heat pump is to be able to compete against other equipment there is still need to optimize components and functions.

Capacity regulation is a special problem, for a heat pump under European climatic conditions will run on partial load for a large part of its operating time. Normally, capacity is controlled by on/off compressor control. This is simple and cheap, but it has a disadvantage in that it increases system losses and increases the risks of creating unsuitable operating conditions.

With the help of computer simulation and based on a "traditional" European heat pump system, this paper describes some of the consequences of using a compressor with adjustable regulation. It also describes a newly-introduced twin compressor suitable for domestic heat pump systems.

DOMESTIC HEAT PUMPS IN NORTHERN EUROPE

Domestic heat pump systems in Northern Europe are almost exclusively linked to heating since air conditioning is seldom seen in private dwellings. "Northern Europe" in this context means, mainly, Sweden, Denmark, West Germany and the Benelux countries. It is in these areas that heat pumps have the widest use at present.

Extremes of temperature in Northern Europe are short-lived. An example is shown in fig. 1. Here the number of hours with heat demand in the course of a year (a Danish reference year) are given. The outside temperatures used to dimension heating systems in the area swing from -10 °C to -18 °C.

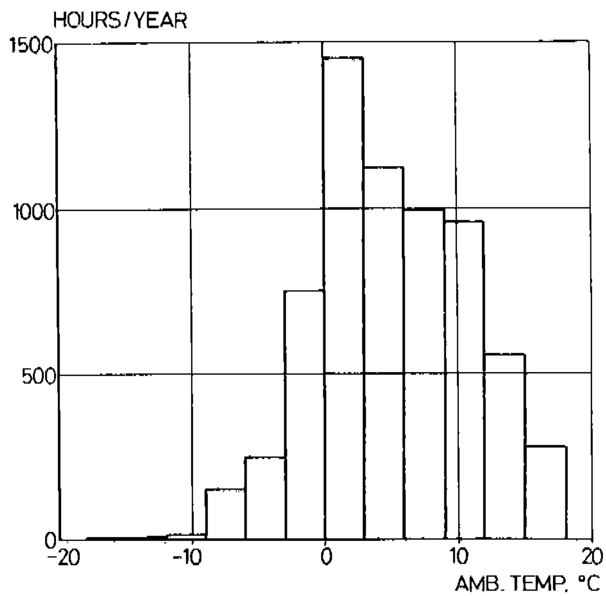

Fig. 1 Temperature distribution in the Danish reference year

Heat distribution systems are normally designed as hydronic or as direct resistance heating. The heat pump systems are mainly hydronic. The heat source can be outside air or water; the latter direct from well bores or in the form of brine from ground coils.

In Scandinavia, water/water systems dominate whereas in the rest of Northern Europe the use of air/water and water/water is equally divided. A predominant use of air/water systems is anticipated in the future.

Water/water systems are often dimensioned to cover the full heating requirement of a dwelling, while air/water systems are often combined with another source of heat to meet heat demands during peak loading (hybrid system). This secondary heat source can be direct using electricity or, frequently, in the case of installation on an existing system, from an oil burner. The secondary heat source can be connected in parallel (parallel hybrid) with the heat pump, or may completely take over the heat supply (alternative hybrid). The two system combinations provide different heat pump operating ranges; parallel operation makes possible very low evaporating temperatures, a requirement that is rare with hybrid alternative operation.

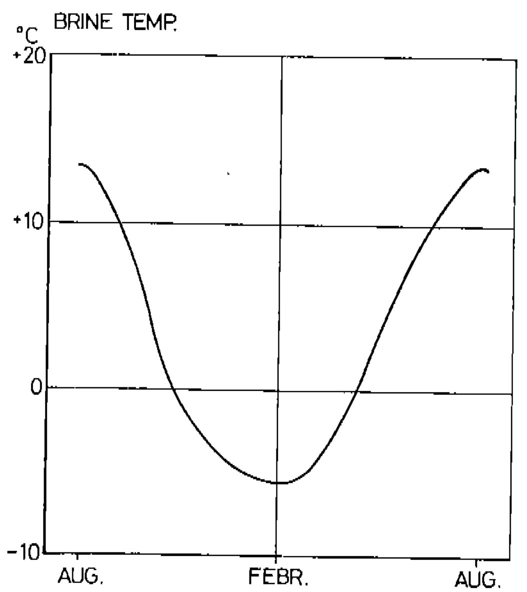

Fig. 2 Brine temperature in a loaded ground coil

The nature of the heat source also affects the extent of the evaporating temperature range. Water from a bore has a constant temperature (8-11 °C) irrespective of the time of the year, whereas the temperature of a ground coil varies. Outside air temperature is of course directly related to the time of year, as is heat demand. Fig. 2 shows a typical distribution of the brine temperature for a loaded ground coil. Fig. 3 shows the yield characteristics for three heat pumps, using the same compressor, for ground water, soil and outside air as heat sources.

Fig. 3 also depicts the average heat requirement for a well-insulated house. Comparison with the heat pump output characteristic shows that only at specific points is there agreement. When outside temperatures are high the output of the heat pump is more than the heat requirement and the pump must run intermittently. With low outside temperatures, heat must be added from a supplementary heat source if the heat requirement is to be met.

The heat demand for new dwellings in Scandinavia will, according to the stipulations laid down by authorities, be 35-50 W/m^2 for an average 130 m^2. For newly-insulated older buildings the requirement will be 70-100 W/m^2. Outside Scandinavia, existing dwellings are usually poorly insulated (70-200 W/m^2) but the future demands of authorities will follow those of Scandinavia. Where the heating of domestic hot (tap) water is integrated in a system, an additional requirement of 0.6 kW must be reckoned on.

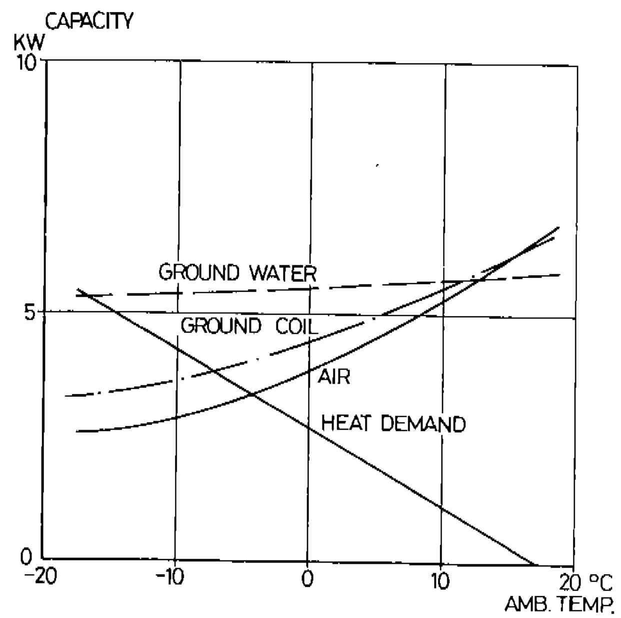

Fig. 3 Heat pump capacity as a function of the heat source

CAPACITY CONTROL

As mentioned in the introduction, the heat capacity of a heat pump is normally simply on/off controlled. An exception is large plant (30 kW or more) with semihermetic compressors equipped with cylinder unloading. On/off control is cheap and reliable but does give some operational disadvantages. Fig. 3 shows that the maximum power requirement is 4.5 kW, whereas a heat pump with outside air as the heat source will yield 8.3 kW to cover this requirement when outside temperatures are high. This yield, in relation to middle and high outside temperatures will not just mean high intermittence, it will severely load the heat exchangers. Both factors reduce system efficiency.

Another problem is presented by unexpected events like the sudden breakdown of fans, circulation pumps, etc. The larger the capacity the faster will the heat pump be exposed to dangerous operating conditions; the electricity supply will more severely loaded than necessary and must therefore be over-dimensioned in relation to the actual demand.

To eliminate these risks or reduce the losses and disadvantages from on/off control, compressor capacity can be made modulating, either in stages or continuously. Several authors have presented solutions and assessed their practical use, e.g. /2/. As regards small and medium size hermetic compressors, the most important possibilities are:

1. Twin compressors
2. Two-speed compressors
3. Frequency-controlled compressors
4. Blocked suction cylinder unloading

Of these proposals, the twin compressor is the most frequently emphasized as being the most suitable when taking into account costs, efficiency and reliability. The frequency-controlled compressor might subsequently become a competitor because of the continued development of better and cheaper power electronic components.

Having said all this however, it must be made clear that capacity-modulating compressors involve increased demands for optimized systems so that improved efficiency does not become absorbed by other losses. For example:

Expansion valves must operate in the double dynamic area. They must be chosen and set with care so that minor temperature differentials across the evaporator are not counteracted by increased superheating of the gas from the evaporator, something that will result in lower suction pressure.

Distribution of refrigerant in the evaporator can become uneven with lower mass flow.

Oil feedback can be compromised.

(Regarding evaporator design, the last two points require much attention).

Auxiliary energy for fans and circulation pump must be carefully related to requirements so that energy saving on the compressor is not counteracted by increased energy for pumps and fans.

Another way of modulating capacity is to build up a heat pump system of parallel-connected, separate part-systems. This avoids the disadvantages mentioned above and affords the advantage of series production. At least one European system manufacturer uses such a module system.

SIMULATION

In order to gain an idea of the most important parameters with the capacity modulation of electrical heat pump systems, three different types of compressors have been compared using a simple simulation programme which describes the basic relations in such systems. The three compressor types are assumed as all having the same maximum yield and performance, corresponding to the compressors described in the following section. See fig. 9. Capacity modulation is considered as displacement or speed reduction without changing the efficiency of the system.

A calculation example with normal capacity modulation is given. This provides a reference for the simulation of a compressor system with two capacity stages, e.g. a twin compressor with two equally large compressor units, or a compressor with a 2/4 pole motor. The third compressor system can be capacity-regulated continuously from 100% to 20% yield, corresponding to what can be achieved with a frequency-controlled induction motor.

Simulation programme

The programme is based on the heat requirement the simulated house will have in a climate corresponding to the Danish reference year, see fig. 1. In fig. 1, no account has been taken of temperatures between 12 and 18 °C in June, July and August because these temperatures occur duting night periods where there is no heating demand.

An air/water system is considered where any added heat is supplied to individual rooms direct in the form of electrical heat independent of the radiator system. The radiator system is designed so that the maximum occurring condensing temperature is approx. 60 °C. The specific heat capacity of the air is assumed constant and defrosting has not been taken account of. Compressor yield and power consumption is determined by characteristics that match calorimeter measurements (fig. 9). The programme does not therefore take account of the expansion valve or the lack of a regenerator. No degradation losses are included. The basic programme is described in detail in /3/.

Calculated results

Three heat pump systems with different heat demands are simulated to find any differences during short and long term operating times. The calculated system conditions for one of these systems is shown in figs. 4, 5 and 6. (The symbols are explained in table I). Not too much weight must be placed on the absolute values because of the much-simplified descriptions of the individual components. Assessments should therefore be restricted to the assessment of trends. Table II gives some of the parameters used.

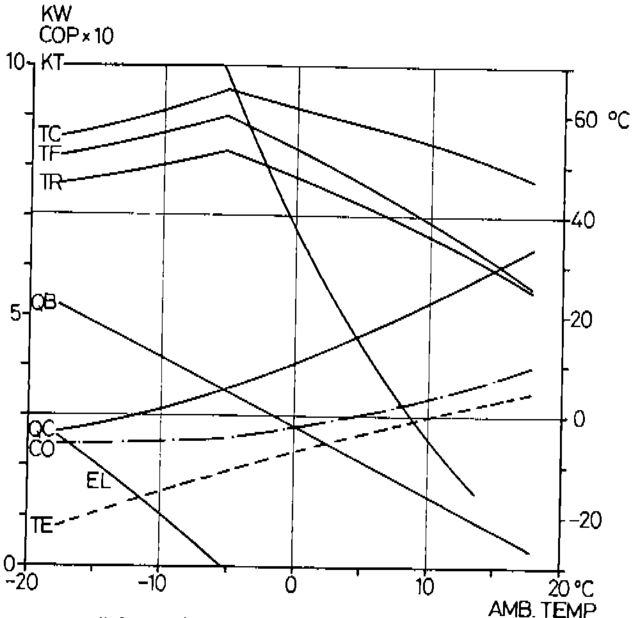

Fig. 4 Single compressor

Fig. 4 shows that there is a balance between demand and heat pump capacity at an ambient temperature of -5 °C, after which the compressor running time falls off as temperature rises. At the same time, the temperature differential across condenser and evaporator rises.

TABLE I

TC = COND. TEMP.	QB = HEAT DEMAND
TE = EVAP. TEMP.	QC = COMPR. CAPACITY
TF = WATER TEMP. (UPSTR)	EL = RESIST. HEAT
TR = WATER TEMP. (DOWNSTR)	KT = DUTY TIME
CO = COMPRESSOR COP	DP = REL. DISPL.

TABLE II

COMPRESSOR:	FIG. 9
SYSTEM: MAX. COND. TEMP.	60°C
AT T-OUT = -5°C: T-EVAP.	-11°C
PUMP. POWER	50W
VENTILATOR POWER	163W
INDOOR TEMP.	20°C
OUTDOOR TEMP.	FIG. 1

Fig. 5 shows a corresponding sequence for a twin compressor system. At ambient temperatures between -5 and +2 °C, one compressor unit runs continuously while the other runs intermittently. KT in fig. 5 is the running time for the intermittent compressor. At temperatures above +2 °C, one of the compressor units is stopped whereby the temperature differential across the heat exchangers becomes halved - compared to operation with both compressor units.

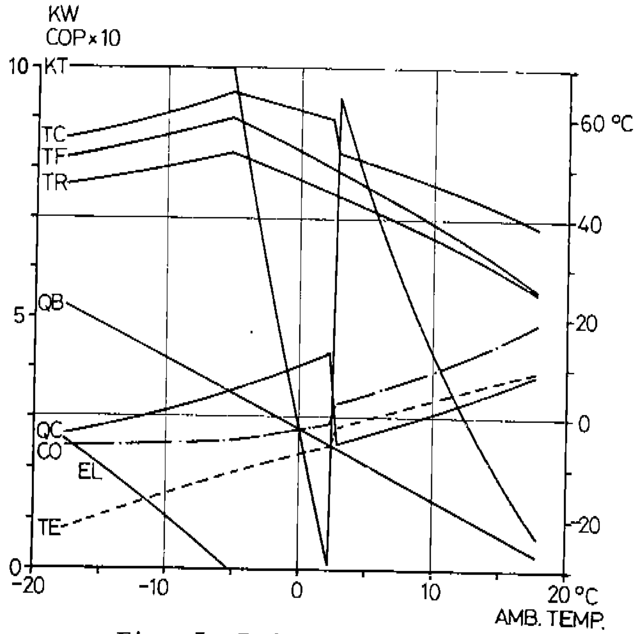

Fig. 5 Twin compressor

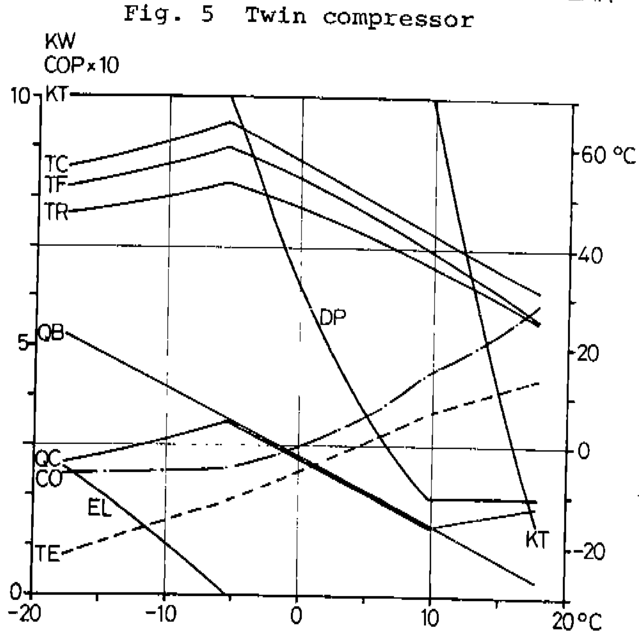

Fig. 6

Fig. 6 shows how the continuous adaptation of compressor capacity to demand minimizes the temperature differential across condenser and evaporator.

All the calculated results are collated in tables III, IV and V. It can be seen that the average compressor efficiency (SCOP) rises markedly with the degree of capacity modulation and the increase with the twin compressor is about 60% of the rise for continuous regulation.

TABLE III

HOUSE: 100 M^2 OF 35 W/M^2 (10.6 MWH/YEAR) HYBRID-POINT AT T-OUT = -10^oC			
COMPRESSOR	SINGLE	TWIN	CONT.
COMPR. POWER KWH	3367	2969	2774
SUPPL. POWER KWH	11	11	11
PUMP. + VENT. POWER KWH	709	961	1251
COMPR. DUTY TIME H	2350	3902	5684
SCOP COMPRESSOR	3.08	3.45	3.69
SCOP SYSTEM	2.60	2.61	2.62

TABLE IV

HOUSE: 125 M^2 OF 35 W/M^2 (13.5 MWH/YEAR) HYBRID-POINT AT T-OUT = -5^oC			
COMPRESSOR	SINGLE	TWIN	CONT.
COMPR. POWER KWH	4494	4134	3931
SUPPL. POWER KWH	115	115	115
PUMP. + VENT. POWER KWH	820	1085	1299
COMPR. DUTY TIME H	3031	4684	5981
SCOP COMPRESSOR	2.94	3.19	3.36
SCOP SYSTEM	2.49	2.54	2.53

TABLE V

HOUSE: 154 M^2 OF 50 W/M^2 (23.5 MWH/YEAR) HYBRID-POINT AT T-OUT = $+2^oC$			
COMPRESSOR	SINGLE	TWIN	CONT.
COMPR. POWER KWH	7010	6734	6651
SUPPL. POWER KWH	2524	2524	2524
PUMP. + VENT. POWER KWH	1093	1257	1364
COMPR. DUTY TIME H	4713	5717	6378
SCOP COMPRESSOR	3.30	3.41	3.43
SCOP SYSTEM	2.21	2.22	2.23

Equally noticeable is that the immediate rise in system efficiency is marginal. This is because the reduction in compressor energy consumption is counteracted by a rise in the energy consumption caused by longer fan running time. This emphasizes that with capacity modulation systems the consumption of auxiliary energy should be looked at and, if possible, controlled.

As previously mentioned, no account has been taken of degradation losses or defrosting losses in these calculations. Both types of losses will be lower in the capacity modulated systems.

With the twin compressor system, degradation losses will only be of signifiance in temperature ranges where one compressor unit runs intermittently and the other unit does not run at all, i.e. at high ambient temperatures with relatively few operating hours.

Degradation losses can only be effectively found by measuring relevant systems and such measuring has not yet been concluded. That is why it is difficult to give precise values. The assumption is however that they are of such a magnitude that even with the above results compressor capacity modulation is an attractive proposition from the point of view of energy saving.

TWIN COMPRESSOR

A capacity-regulated heat pump compressor with a rated heat output of 5.5 kW for use in smaller well-insulated dwellings and larger buildings with hybrid systems or module systems has been developed and introduced. The compressor is designed as a twin compressor based on two compressors of the same size in separate shells (fig. 7).

The compressor is fitted with a common external suction muffler to ensure uniform loading of the two units. Correspondingly, the oil sumps of the two units are connected with a crankcase pressure equalizing tube so that the oil level in them is always the same. The discharge connectors are lead out separately and have to be fitted with checkvalves. This is necessary to avoid condensation in the one unit immediately after a cold start on the other unit. The checkvalves also lessen starting torque requirements on the stationary unit when the other unit is running.

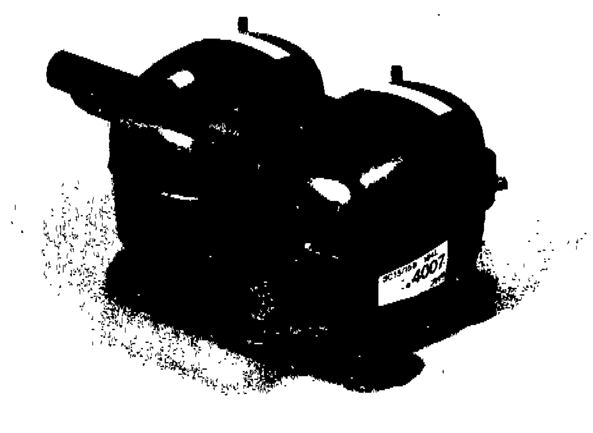

Fig. 7 Twin compressor

The compressor is oil-cooled via a tube coil in both oil sumps. The oil cooler will normally be connected direct to the radiator outlet (see fig. 8) but can also be cooled with pressure gas via a line from the condenser. The use of such an oil cooler means that the compressor can be completely insulated so that all compressor power loss can be effectively transferred to the heat receiver. Further, there is no risk of extreme temperatures in the compressor when suction pressure is abnormally low.

The oil cooler also ensures that during compressor standstill oil is held at the temperature of the radiator water, i.e. from 35 to 55 °C. This avoids refrigerant absorption in the oil and thereby eliminates the risk of slugging.

The compressor is equipped with two single-phase PSC motors with start capacitors which enable users to choose single or three-phase mains installation. The starting torque is sufficient for one unit to start while the other unit is running. On demand for full capacity, e.g. after defrosting, the start of one unit while the other unit has carried out a start is delayed in order to minimize start surge in the electrical wiring.

The compressor is specially optimized for heat pump applications. For one thing, it is designed for an evaporating temperature range -10 to +5 °C (see fig. 10) within which it will run for most of its operating time. Likewise, the valve and muffler system is optimized and the bearing function has been thouroughly examined.

For the manufacturers of the compressor, the decisive factors in choosing to use two separate shells instead of a common shell were:

Shorter development time

Lower noise level

Use of standard components

As opposed to this, a common shell would have meant:

Less volume

Fewer components in production

The performance of the twin compressor with both units cut in is given in fig. 9. With one unit stopped, yield and power consumption are halved. The heat output is the sum of the measured evaporator output and power consumption. This is a correct expression of heat output since during normal use the compressor is insulated and there are no losses to the surroundings.

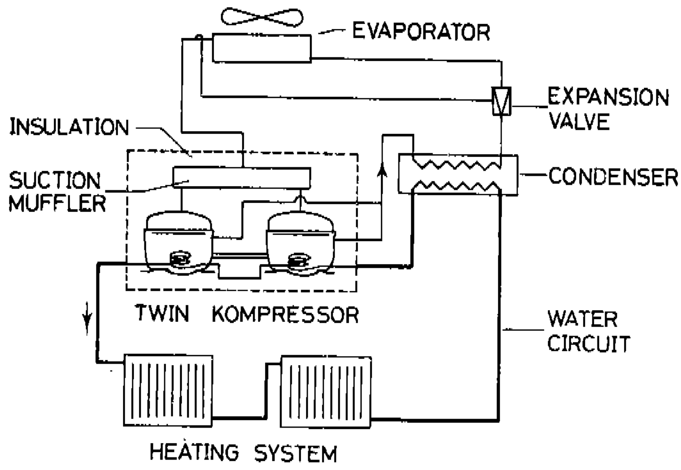

Fig. 8

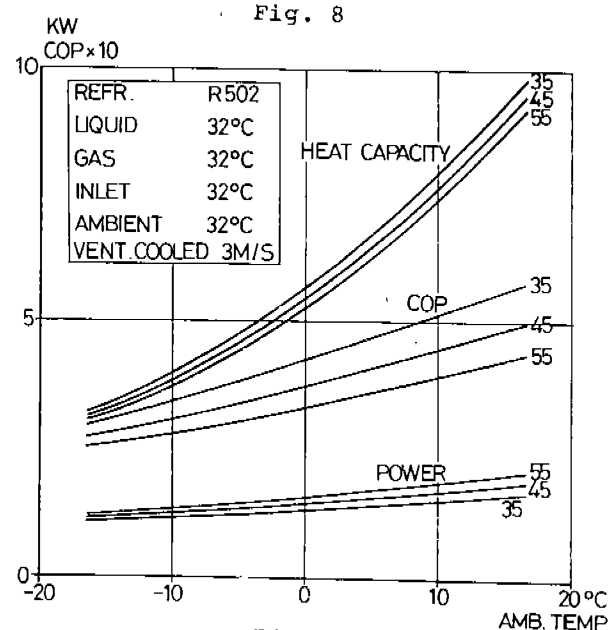

Fig. 9

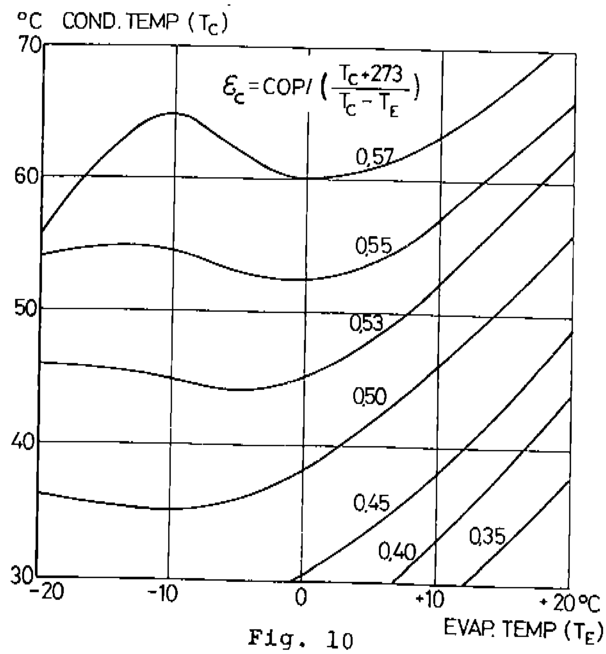

Fig. 10

SUMMARY

Using typical European heat pump systems as a background, two types of capacity modulated compressors are compared with the pure on/off controlled traditional compressor, using simulation under simplified conditions. Depending on how large a part of the total heat demand the heat pump must cover, the average seasonal efficiency of the compressor rises 5 to 20% with infinite capacity regulation. Modulation in two stages produces 60% of this increase.

Under simulated conditions, without fan output modulation and without taking account of defrosting and degradation losses, the rise in system efficiency is marginal.

As far as energy and operation are concerned however, compressor capacity modulation is attractive - assuming that the system is optimized correctly in view of the importance of degradation losses.

As an example of a capacity-regulated heat pump compressor, a twin compressor is separately described.

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REFERENCES

1. Cube, H.L. von (1977): Heat Sources for Heat Pumps, Proc. of 1977 Wärmepumpen Tagung in Essen.
2. Scharf, Ph.H; King, R.W.; Yu, J.D. (1978): Comparison of Capacity Modulation Means for Hermetic Compressors, Proc. of the 1978 Purdue Compressor Technology Conference.
3. Danig, P.O. (1977): Sammenligning af luft/luft og luft/radiator varmepumpe ved simulering, Scandinavian Refrigeration 1/77.